Chapter 5: Energy Penalties of Cooling Towers

INTRODUCTION

For the Existing Facility 316(b) Proposal, the Agency considered regulatory options in which regulated facilities (or a subset thereof) would achieve flow reduction commensurate with closed-cycle wet cooling systems. In addition, the Agency analyzed regulatory options based on flow reduction commensurate with near-zero intake of dry cooling systems. This chapter discusses the topics of energy penalties of such cooling systems.

For the Section 316(b) New Facility Final Rule the Agency researched and derived energy penalty estimates, based on empirical data and proven theoretical concepts, for a variety of conditions. The regulatory analysis conducted by the Agency for this Existing Facility Section 316(b) Proposal utilized the results of the New Facility analysis. This chapter presents the research, methodology, results, and conclusions for the Agency's thorough effort to estimate energy penalties due to the operational performance of power plant cooling systems.

As a consequence of energy penalties for some cooling systems, increased air pollutant emissions may occur for some power plants as compared to a baseline system. The discussion of air pollutant emissions and other potential environmental impacts from cooling towers are presented in Chapter 6 of this document.

The remainder of this chapter is organized as follows:

- Section 5.1 presents the energy penalty estimates used for analysis of the flow reduction regulatory options.
- Section 5.2 presents an introduction to the Agency's energy penalty estimates.
- Section 5.3 focuses on steam turbines and the changes in efficiency associated with using alternative cooling systems.
- Section 5.4 evaluates the net difference in required pumping and fan energy for alternative cooling systems.
- Section 5.5 combines and summarizes the energy impacts of pumping and fan energy requirements for alternative cooling systems.
- Section 5.6 summarizes data from other sources on the potential energy penalty of alternative cooling systems at existing facilities.

5.1 Energy Penalty Estimates for Cooling

Tables 5-1 through 5-4 present the energy penalty estimates utilized for assessing the operational energy impacts of certain, flow-reduction regulatory options considered for this proposal. The Agency presents the methodology for estimation of energy penalties in Sections 5.2 through 5.5 of this chapter.

Table 5-1: National Average, Mean-Annual Energy Penalty, Summary Table

Cooling Type	Percent Maximum Load[a]	Mean-Annual Nuclear Percent of Plant Output	Mean-Annual Combined-Cycle Percent of Plant Output	Mean-Annual Fossil-Fuel Percent of Plant Output
Wet Tower vs. Once-Through	67	1.7	0.4	1.7
Dry Tower vs. Once-Through	67	8.5	2.1	8.6
Dry Tower vs. Wet Tower	67	6.8	1.7	6.9

[a] For calculating the average annual penalties, the Agency conservatively estimated that plants will operate over the course of the year at non-peak loads. See below for a discussion of percent maximum load.

Table 5-2: National Average, Peak-Summer Energy Penalty, Summary Table

Cooling Type	Percent Maximum Load[a]	Peak-Summer Nuclear Percent of Plant Output	Peak-Summer Combined-Cycle Percent of Plant Output	Peak-Summer Fossil-Fuel Percent of Plant Output
Wet Tower vs. Once-Through	100	1.9	0.4	1.7
Dry Tower vs. Once-Through	100	11.4	2.8	10.0
Dry Tower vs. Wet Tower	100	9.6	2.4	8.4

[a] Peak-summer shortfalls occur when plants are at or near maximum capacity.

The Agency developed its estimates of average annual energy penalties based on the assumption that during non-peak loads turbines would operate at roughly 67 percent of maximum peak load. Therefore, the Agency's estimates of annual energy penalties in Tables 5-1 and 5-3 represent calculations of turbine energy penalties at 67 percent of maximum load. The Agency considered this to be a conservative assumption for the calculation of energy penalties because turbine efficiency is considerably higher for the 100 percent of maximum load condition. The Agency understands, based on discussions with the Department of Energy, that a significant portion of existing power plants, when dispatched, would likely operate at near maximum loads. Therefore, the turbine energy penalty portion of mean annual energy penalty estimates presented in Tables 5-1 and 5-3 could be overstated. The Agency estimates that had it calculated the mean annual penalties for the 100 percent of maximum load condition, the national average annual energy penalty of wet cooling versus once-through systems would be approximately 0.3 percent for combined-cycle, 1.1 percent for fossil-fuel, and 1.3 for nuclear plants. However, the Agency utilized the higher values in Tables 5-1 and 5-3 for the economic analyses of the regulatory options considered for this proposal.

Table 5-3: Total Energy Penalties at 67 Percent Maximum Load[a]

Location	Cooling Type	Nuclear Annual Average	Combined-Cycle Annual Average	Fossil-Fuel Annual Average
Boston	Wet Tower vs. Once-Through	1.6	0.4	1.6
	Dry Tower vs. Once-Through	7.4	1.8	7.1
	Dry Tower vs. Wet Tower	5.8	1.4	5.5
Jacksonville	Wet Tower vs. Once-Through	1.9	0.4	1.7
	Dry Tower vs. Once-Through	12.0	3.0	12.5
	Dry Tower vs. Wet Tower	10.1	2.5	10.8
Chicago	Wet Tower vs. Once-Through	1.8	0.4	1.8
	Dry Tower vs. Once-Through	7.8	1.9	7.7
	Dry Tower vs. Wet Tower	5.9	1.5	5.9
Seattle	Wet Tower vs. Once-Through	1.5	0.4	1.5
	Dry Tower vs. Once-Through	7.0	1.7	6.9
	Dry Tower vs. Wet Tower	5.5	1.3	5.4

[a] For calculating the average annual penalties, the Agency conservatively estimated that plants will operate over the course of the year at non-peak loads. See above for a discussion of percent maximum load.

Table 5-4: Total Energy Penalties at 100 Percent Maximum Load[a]

Location	Cooling Type	Peak-Summer Nuclear Percent of Plant Output	Peak-Summer Combined-Cycle Percent of Plant Output	Peak-Summer Fossil-Fuel Percent of Plant Output
Boston	Wet Tower vs. Once-Through	2.1	0.5	1.9
	Dry Tower vs. Once-Through	11.6	2.9	10.2
	Dry Tower vs. Wet Tower	9.5	2.4	8.3
Jacksonville	Wet Tower vs. Once-Through	1.6	0.4	1.4
	Dry Tower vs. Once-Through	12.3	3.1	10.7
	Dry Tower vs. Wet Tower	10.7	2.7	9.3
Chicago	Wet Tower vs. Once-Through	2.2	0.5	2.0
	Dry Tower vs. Once-Through	11.9	2.9	10.4
	Dry Tower vs. Wet Tower	9.6	2.4	8.4
Seattle	Wet Tower vs. Once-Through	1.6	0.4	1.5
	Dry Tower vs. Once-Through	10.0	2.4	8.9
	Dry Tower vs. Wet Tower	8.4	2.0	7.4

[a] Peak-summer shortfalls occur when plants are at or near maximum capacity.

5.2 Introduction to Energy Penalty Estimates

This energy penalty discussion presents differences in steam power plant efficiency or output associated with the effect of using alternative cooling systems. In particular, this evaluation focuses on power plants that use steam turbines and the changes in efficiency associated with using alternative cooling systems. The cooling systems evaluated include: once-through cooling systems; wet tower closed-cycle systems; and direct-dry cooling systems using air cooled condensers. However, the methodology is flexible and can be extended to other alternative types of cooling systems so long as the steam condenser performance or the steam turbine exhaust pressure can be estimated.

The energy penalties presented in this chapter were developed for new, "greenfield" facilities. As such, the Agency estimates for this proposal for existing facilities that the energy penalties of cooling system conversions from once-through to recirculating wet cooling towers would be similar to the new, "greenfield" cases. The Department of Energy expressed concern that this methodology may underestimate the pumping energy requirements of recirculating wet tower systems for converted cooling systems. This matter, among others, is discussed in Section 5.6 below.

The Agency acknowledges that direct-dry cooling systems are unlikely candidates for cooling system conversions at existing power plants. A direct-dry cooling system (as discussed in Appendix D of this document) condenses the exhaust steam that is fed directly to the dry tower from the generating turbine. However, steam turbines at the existing power plants within the scope of this rule are, without exception, configured to condense steam utilizing a surface condenser system. Therefore, the only type of dry cooling system that would be considered for a cooling system conversion is an indirect-air cooled condenser. Otherwise, the entire steam turbine would be replaced or dramatically reconfigured to feed exhaust steam to a direct-dry cooling system. The engineering feasibility of this type of plant reconfiguration was considered unproven by the Agency and the costs of turbine replacement were also deemed too high for this proposal.

Indirect-dry cooling systems operate less efficiently than direct-dry cooled systems. Therefore, the energy penalties for dry cooling systems presented in this chapter would be higher for the only application that would be considered for existing facilities. The Department of Energy (DOE) studied the peak summer energy penalty resulting from converting plants with once-through cooling to wet towers or indirect dry towers (see DCN 4-2512). DOE modeled five locations – Delaware River Basin (Philadelphia), Michigan/Great Lakes (Detroit), Ohio River Valley (Indianapolis), South (Atlanta), and Southwest (Yuma) – using an ASPEN simulator model. The model evaluated the performance and energy penalty for hypothetical 400-MW coal-fired plants that were retrofitted from using once-through cooling systems to wet- and dry-recirculating systems. The DOE estimates that conversion to an indirect-dry tower could cause peak summer energy penalties ranging from 8.9 percent to 14.1 percent with a design approach of 20 degrees Fahrenheit and 12.7 percent to approximately 18 percent with an approach of 40 degrees Fahrenheit. Note that these peak summer energy penalties are higher than those estimated by EPA (as presented in Tables 5-2 and 5-4 above) for the direct-dry cooling system. The Agency's estimates of direct-dry cooling system peak summer energy penalties range from 7.4 percent to 10.7 percent for fossil-fuel plants. As such, the analysis of energy effects of the dry cooling-based regulatory options considered for this proposal may not reflect the full magnitude of the energy penalty of the indirect-dry cooling systems.

5.2.1 Power Plant Efficiencies

Most power plants that use a heat-generating fuel as the power source use a steam cycle referred to as a "Rankine Engine," in which water is heated into steam in a boiler and the steam is then passed through a turbine (Woodruff 1998). After exiting the turbine, the spent steam is condensed back into water and pumped back into the boiler to repeat the cycle. The turbine, in turn, drives a generator that produces electricity. As with any system that converts energy from one form to another, not all of the energy available from the fuel source can be converted into useful energy in a power plant.

Steam turbines extract power from steam as the steam passes from high pressure and high temperature conditions at the turbine inlet to low pressure and lower temperature conditions at the turbine outlet. Steam exiting the turbine goes to the condenser, where it is condensed to water. The condensation process is what creates the low pressure conditions at the turbine outlet. The steam turbine outlet or exhaust pressure (which is often a partial vacuum) is a function of the temperature maintained at the condensing surface (among other factors) and the value of the exhaust pressure can have a direct effect on the energy available to drive the turbine. The lower the exhaust pressure, the greater the amount of energy that is available to drive the turbine, which in turn increases the overall efficiency of the system since no additional fuel energy is involved.

The temperature of the condensing surface is dependent on the design and operating conditions within the condensing system (e.g., surface area, materials, cooling fluid flow rate, etc.) and especially the temperature of the cooling water or air used to absorb heat and reject it from the condenser. Thus, the use of a different cooling system can affect the temperature maintained at the steam condensing surface (true in many circumstances). This difference can result in a change in the efficiency of the power plant. These efficiency differences vary throughout the year and may be more pronounced during the warmer months. Equally important is the fact that most alternative cooling systems will require a different amount of power to operate equipment such as fans and pumps, which also can have an effect on the overall plant energy efficiency. The reductions in energy output resulting from the energy required to operate the cooling system equipment are often referred to as parasitic losses.

In general, the penalty described here is only associated with power plants that utilize a steam cycle for power production. Therefore, this analysis will focus only on steam turbine power plants and combined-cycle gas plants. The most common steam turbine power plants are those powered by steam generated in boilers heated by the combustion of fossil fuels or by nuclear reactors.

Combined-cycle plants use a two-step process in which the first step consists of turbines powered directly by high pressure hot gases from the combustion of natural gas, oil, or gasified coal. The second step consists of a steam cycle in which a turbine is powered by steam generated in a boiler heated by the low pressure hot gases exiting the gas turbines. Consequently, the combined-cycle plants have much greater overall system efficiencies. However, the energy penalty associated with using alternative cooling systems is only associated with the steam cycle portion of the system. Because steam plants cannot be quickly started or stopped, they often operate as base load plants, which are continuously run to serve the minimum load required by the system. Since combined-cycle plants obtain only a portion of their energy from the slow-to-start/stop steam power step, the inefficiency of the start-up/stop time period is more economically acceptable and therefore they are generally used for intermediate loads. In other words, they are started and stopped at a greater frequency and with greater efficiency than base load steam plant facilities.

One measure of the plant thermal efficiency used by the power industry is the Net Plant Heat Rate (NPHR), which is the ratio of the total fuel heat input (BTU/hr) divided by the net electric generation (kW). The net electric generation includes only electricity that leaves the plant. The total energy plant efficiency can be calculated from the NPHR using the following formula:

$$\text{Plant Energy Efficiency} = 3413 / \text{NPHR} \times 100 \qquad (1)$$

Table 5-5 presents the NPHR and plant efficiency numbers for different types of power plants. Note that while there may be some differences in efficiencies for steam turbine systems using different fossil fuels, these differences are not significant enough for consideration here. The data presented to represent fossil fuel plants is for coal-fired plants, which comprise the majority in that category.

Table 5-5: Heat Rates and Plant Efficiencies for Different Types of Steam Powered Plants

Type of Plant	Net Plant Heat Rate (BTU/kWh)	Efficiency (%)
Steam Turbine - Fossil Fuel	9,355	37 to 40
Steam Turbine – Nuclear	10,200	34
Combined Cycle – Gas	6,762	51
Combustion Turbine	11,488	30

Source: Analyzing Electric Power Generation under the CAAA. Office of Air and Radiation U.S. Environmental Protection Agency. April 1996 (Projections for year 2000-2004).

Overall, fossil fuel steam electric power plants have net efficiencies with regard to the available fuel heat energy ranging from 37 to 40 percent. Attachment A at the end of this chapter (Ishigai, S. 1999.) shows a steam power plant heat diagram in which approximately 40 percent of the energy is converted to the power output and 44 percent exits the system through the condensation of the turbine exhaust steam, which exits the system primarily through the cooling system with the remainder exiting the system through various other means including exhaust gases. Note that the exergy diagram in Attachment A shows that this heat passing through the condenser is not a significant source of plant inefficiency, but as would be expected it shows a similar percent of available energy being converted to power as shown in Table 5-5 and Attachment A.

Nuclear plants have a lower overall efficiency of approximately 34 percent, due to the fact that they generally operate at lower boiler temperatures and pressures and the fact that they use an additional heat transfer loop. In nuclear plants, heat is extracted from the core using a primary loop of pressurized liquid such as water. The steam is then formed in a secondary boiler system. This indirect steam generation arrangement results in lower boiler temperatures and pressures, but is deemed necessary to provide for safer operation of the reactor and to help prevent the release of radioactive substances. Nuclear reactors generate a near constant heat output when operating and therefore tend to produce a near constant electric output.

Combustion turbines are shown here for comparative purposes only. Combustion turbine plants use only the force of hot gases produced by combustion of the fuel to drive the turbines. Therefore, they do not require much cooling water since they do not use steam in the process, but they are also not as efficient as steam plants. They are, however, more readily able to start and stop quickly and therefore are generally used for peaking loads.

Combined cycle plants have the highest efficiency because they combine the energy extraction methods of both combustion turbine and steam cycle systems. Efficiencies as high as 58 percent have been reported (Woodruff 1998). Only the efficiency of the second stage (which is a steam cycle) is affected by cooling water temperatures. Therefore, for the purposes of this analysis, the energy penalty for combined cycle plants is applicable only to the energy output of the steam plant component, which is generally reported to be approximately one-third of the overall combined-cycle plant energy output.

5.3 Turbine Efficiency Energy Penalty

5.3.1 Effect of Turbine Exhaust Pressure

The temperature of the cooling water (or air in air-cooled systems) entering the steam cycle condensers affects the exhaust pressure at the outlet of the turbine. In general, a lower cooling water or air temperature at the condenser inlet will result in a lower turbine exhaust pressure. Note that for a simple steam turbine, the available energy is equal to the difference in the enthalpy of the inlet steam and the combined enthalpy of the steam and condensed moisture at the turbine outlet. A reduction in the outlet steam pressure results in a lower outlet steam enthalpy. A reduction in the enthalpy of the turbine exhaust steam, in combination with an increase in the partial condensation of the steam, results in an increase in the efficiency of the turbine system. Of course, not all of this energy is converted to the torque energy (work) that is available to turn the generator, since steam and heat flow through the turbine systems is complex with various losses and returns throughout the system.

The turbine efficiency energy penalty as described below rises and drops in direct response to the temperature of the cooling water (or air in air-cooled systems) delivered to the steam plant condenser. As a result, it tends to peak during the summer and may be substantially diminished or not exist at all during other parts of the year.

The design and operation of the steam condensing system can also affect the system efficiency. In general, design and operational changes that improve system efficiency such as greater condenser surface areas and coolant flow rates will tend to result in an increase in the economic costs and potentially the environmental detriments of the system. Thus, the design and operation of individual systems can differ depending on financial decisions and other site-specific conditions. Consideration of such site-specific design variations is beyond the scope of this evaluation. Therefore, conditions that represent a typical, or average, system derived from available information for each technology will be used. However, regional and annual differences in cooling fluid temperatures are considered. Where uncertainty exists, a conservative estimate is used. In this context, conservative means the penalty estimate is biased toward a higher value.

Literature sources indicate that condenser inlet temperatures of 55 °F and 95 °F will produce turbine exhaust pressures of 1.5 and 3.5 inches Hg, respectively, in a typical surface condenser (Woodruff 1998). If the turbine steam inlet conditions remain constant, lower turbine exhaust pressures will result in greater changes in steam enthalpy between the turbine inlet and outlet. This in turn will result in higher available energy and higher turbine efficiencies.
The lower outlet pressures can also result in the formation of condensed liquid water within the low pressure end of the turbine. Note that liquid water has a significantly lower enthalpy value which, based on enthalpy alone, should result in even greater turbine efficiencies. However, the physical effects of moisture in the turbines can cause damage to the turbine blades and can result in lower efficiencies than would be expected based on enthalpy data alone. This damage and lower efficiency is due to the fact that the moisture does not follow the steam path and impinges upon the turbine blades. More importantly, as the pressure in the turbine drops, the steam volume increases. While the turbines are designed to accommodate this increase in volume through a progressive increase in the cross-sectional area, economic considerations tend to limit the size increase such that the turbine cannot fully accommodate the expansion that occurs at very low exhaust pressures.

Thus, for typical turbines, as the exhaust pressure drops below a certain level, the increase in the volume of the steam is not fully accommodated by the turbine geometry, resulting in an increase in steam velocity near the turbine exit. This increase in steam velocity results in the conversion of a portion of the available steam energy to kinetic energy, thus reducing the energy that could otherwise be available to drive the turbine. Note that kinetic energy is proportional to the square of the velocity. Consequently, as the steam velocity increases, the resultant progressive reduction in available energy tends to offset the gains in available energy that would result from the greater enthalpy changes due to the reduced pressure. Thus, the expansion of the steam within the turbine and the formation of condensed moisture establishes a practical lower limit for turbine exhaust pressures, reducing the efficiency advantage of even lower condenser surface temperatures particularly at higher turbine steam loading rates. As can be seen in the turbine performance curves presented below, this reduction in efficiency at lower exhaust pressures is most pronounced at higher turbine steam loading rates. This is due to the fact that higher steam loading rates will produce proportionately higher turbine exit velocities.

Attachment B presents several graphs showing the change in heat rate resulting from differences in the turbine exhaust pressure at a nuclear power plant, a fossil fueled power plant, and a combined-cycle power plant (steam portion). The first graph (Attachment B-1) is for a GE turbine and was submitted by the industry in support of an analysis for a nuclear power plant. The second graph (Attachment B-2) is from a steam turbine technical manual and is for a turbine operating at steam temperatures and pressures consistent with a sub-critical fossil fuel plant (2,400 psig, 1,000 °F). The third graph (Attachment B-3) is from an engineering report analyzing operational considerations and design of modifications to a cooling system for a combined-cycle power plant.

The changes in heat rate shown in the graphs can be converted to changes in turbine efficiency using Equation 1. Several curves on each graph show that the degree of the change (slope of the curve) decreases with increasing loads. Note that the amount of electricity being generated will also vary with the steam loading rates such that the more pronounced reduction in efficiency at lower steam loading rates applies to a reduced power output. The curves also indicate that, at higher steam loads, the plant efficiency optimizes at an exhaust pressure of approximately 1.5 inches Hg. At lower exhaust pressures the effect of increased steam velocities actually results in a reduction in overall efficiency. The graphs in Attachment B will serve as the basis for estimating the energy penalty for each type of facility.

Since the turbine efficiency varies with the steam loading rate, it is important to relate the steam loading rates to typical operating conditions. It is apparent from the heat rate curves in Attachment B that peak loading, particularly if the exhaust pressure is close to 1.5 inches Hg, presents the most efficient and desirable operating condition. Obviously, during peak loading periods, all turbines will be operating near the maximum steam loading rates and the energy penalty derived from the maximum loading curve would apply. It is also reasonable to assume that power plants that operate as base load facilities will operate near maximum load for a majority of the time they are operating. However, there will be times when the power plant is not operating at peak capacity. One measure of this is the capacity factor, which is the ratio of the average load on the plant over a given period to its total capacity. For example, if a 200 MW plant operates, on average, at 50 percent of capacity (producing an average of 100 MW when operating) over a year, then its capacity factor would be 50 percent.

The average capacity factor for nuclear power plants in the U.S. has been improving steadily and recently has been reported to be approximately 89 percent. This suggests that for nuclear power plants, the majority appear to be operating near capacity most of the time. Therefore, use of the energy penalty factors derived from the maximum load curves for nuclear power plants is reasonably valid. In 1998, utility coal plants operated at an average capacity of 69 percent (DOE 2000). Therefore, use of the energy penalty values derived from the 67 percent load curves would appear to be more appropriate for fossil-fuel plants. Capacity factors for combined-cycle plants tend to be lower than coal-fired plants and use of the energy penalty values derived from the 67 percent load curves rather than the 100 percent load curves would be appropriate.

5.3.2 Estimated Changes in Turbine Efficiency

Table 5-6 below presents a summary of steam plant turbine inlet operating conditions for various types of steam plants described in literature. EPA performed a rudimentary estimation of the theoretical energy penalty based on steam enthalpy data using turbine inlet conditions similar to those shown in Table 5-6. EPA found that the theoretical values were similar to the changes in plant efficiency derived from the changes in heat rate shown in Attachment B. The theoretical calculations indicated that the energy penalties for the two different types of fossil fuel plants (sub-critical and super-critical) were similar in value, with the sub-critical plant having the larger penalty. Since the two types of fossil fuel plants had similar penalty values, only one was selected for use in the analysis in order to simplify the analysis. The type of plant with the greater penalty value (i.e., sub-critical fossil fuel) was selected as representative of both types.

Table 5-6: Summary of Steam Plant Operating Conditions from Various Sources

System Type	Inlet Temp. / Pressure	Outlet Pressure	Comments	Source
Fossil Fuel - Sub-critical Recirculating Boiler	Not Given / 2,415 psia	1.5 In. Hg	Large Plants (>500MW) have three (high, med, low) pressure turbines. Reheated boiler feed water is 540 °F.	Kirk-Othmer 1997
Fossil Fuel - Super-critical Once-through Boiler	1,000 °F / 3,515 psia	Not Given		Kirk-Othmer 1997
Nuclear	595 °F / 900 psia	2.5 In. Hg	Plants have two (high, low) pressure turbines with low pressure turbine data at left. Reheated boiler feed water is 464 °F.	Kirk-Othmer 1997
Combined Cycle	Gas - 2,400 °F Steam - 900 °F	Not Given	Operating efficiency ranges from 45-53%	www.greentie.org
Fossil Fuel Ranges	900-1,000 °F / 1,800-3,600 psia	1.0-4.5 In Hg	Outlet pressures can be even higher with high cooling water temperatures or air cooled condensers.	Woodruff 1998.

The three turbine performance curve graphs in Attachment B present the change in heat rate from which changes in plant efficiency were calculated. The change in heat rate value for several points along each curve was determined and then converted to changes in efficiency using Equation 1. The calculated efficiency values derived from the Attachment B graphs representing the 100 percent or maximum steam load and the 67 percent steam load conditions have been plotted in Figure 1. Curves were then fitted to these data to obtain equations that can be used to estimate energy penalties. Figure 1 establishes the energy efficiency and turbine exhaust pressure relationship. The next step is to relate the turbine exhaust pressure to ambient conditions and to determine ambient conditions for selected locations.

Note that for fossil fuel plants the energy penalty affects mostly the amount of fuel used, since operating conditions can be modified, within limits, to offset the penalty. However, the same is not true for nuclear plants, which are constrained by the limitations of the reactor system.

Figure 1
Plot of Various Turbine Exhaust Pressure Correction Curves for 100% and 67% Steam Loads

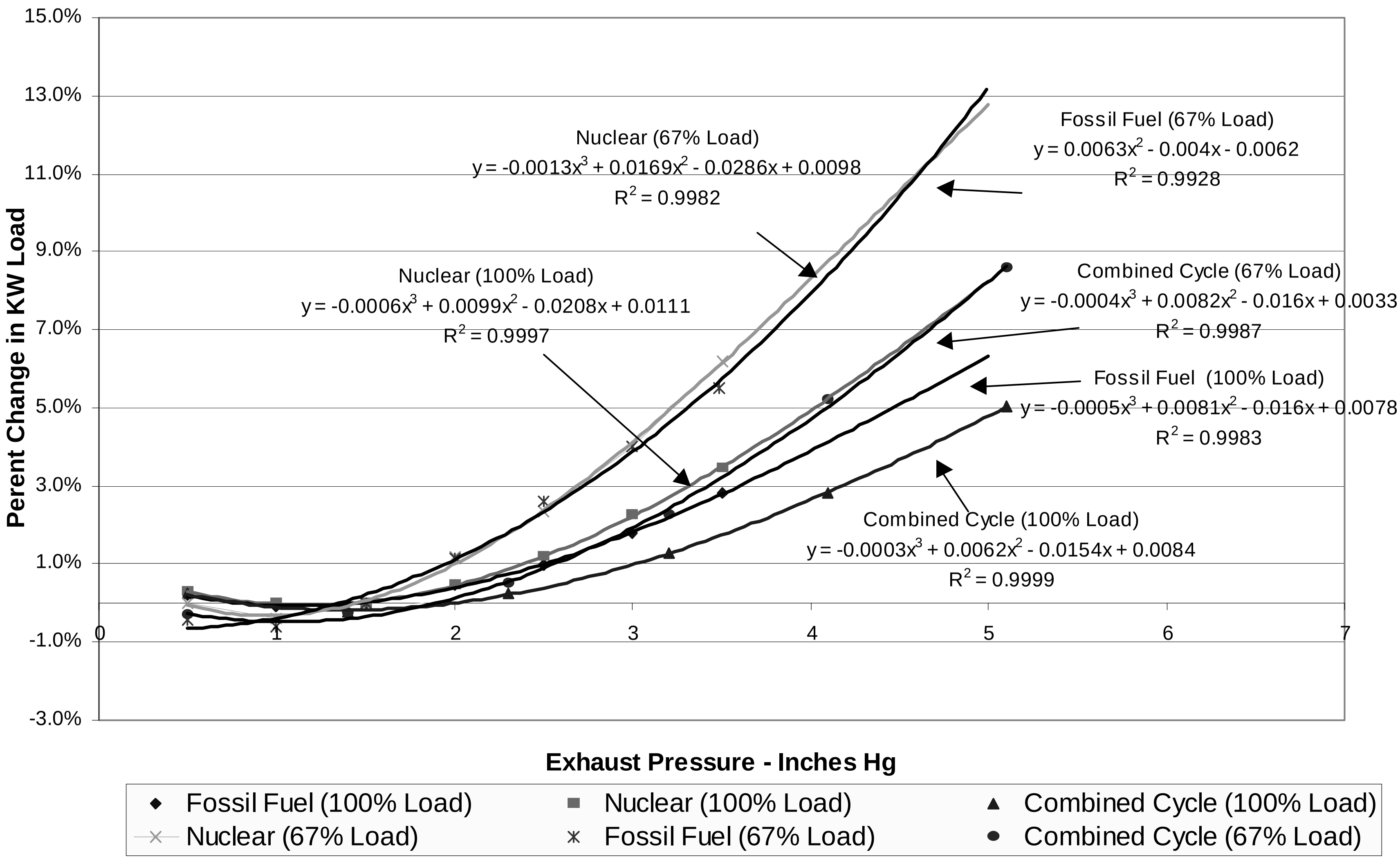
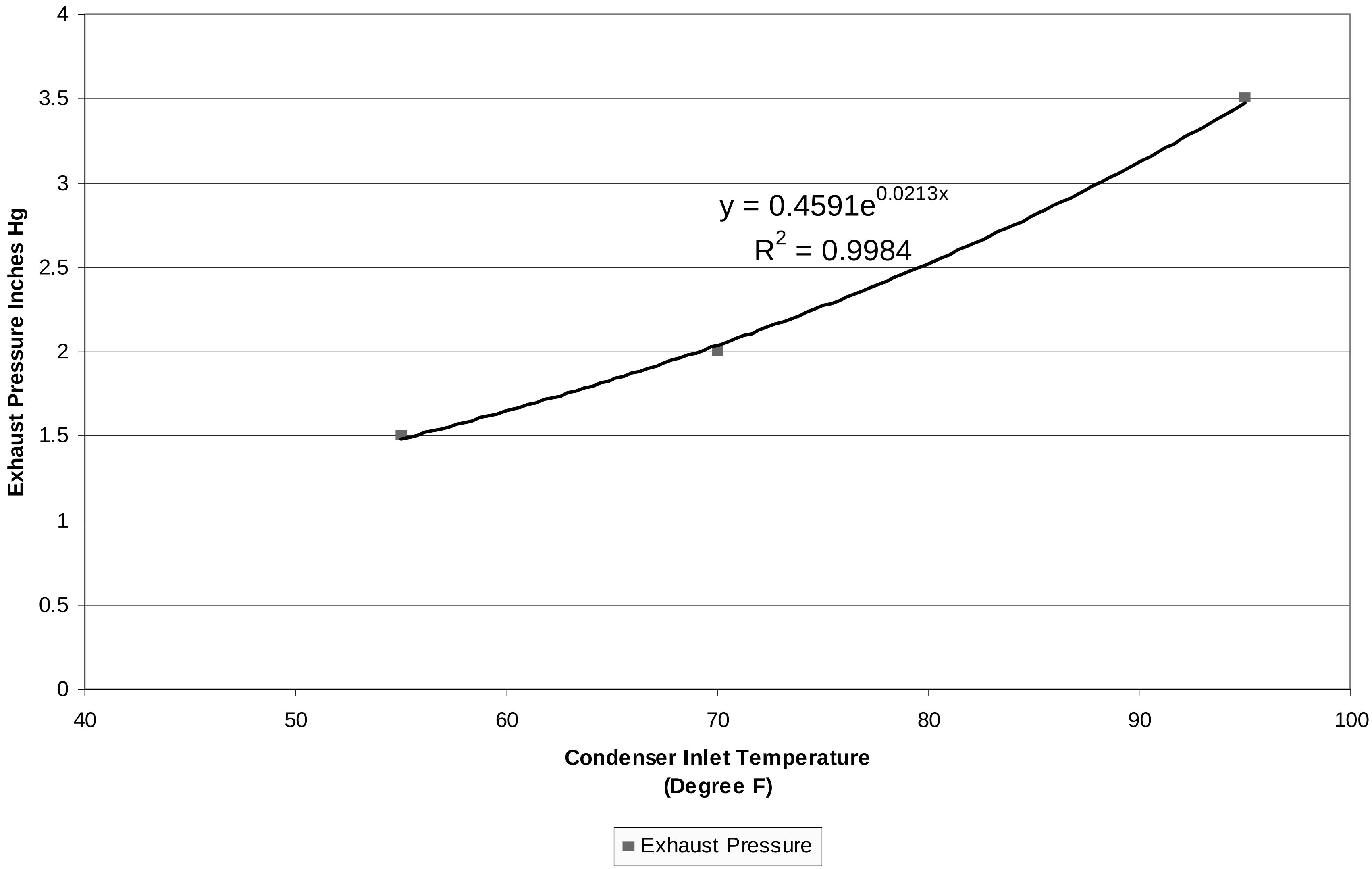


Figure 2
Surface Condenser Cooling Water Inlet Temperature and Steam Pressure Relationship



.3.3 Relationship of Condenser Cooling Water (or Air) Temperature to Steam Side Pressure for Different Cooling System Types and Operating Conditions

❖ ***Surface Condensers***

Both once-through and wet cooling towers use surface condensers. As noted previously, condenser inlet temperatures of 55 °F and 95 °F will produce turbine exhaust pressures of 1.5 and 3.5 inches Hg, respectively. Additionally, data from the Calvert Cliffs nuclear power plant showed an exhaust pressure of 2.0 inches Hg at a cooling water temperature of 70 °F. Figure 2 provides a plot of these data which, even though they are from two sources, appear to be consistent. A curve was fitted to these data and was used as the basis for estimating the turbine exhaust pressure for different surface condenser cooling water inlet temperatures. Note that this methodology is based on empirical data that simplifies the relationship between turbine exhaust pressure and condenser inlet temperature, which would otherwise require more complex heat exchange calculations. Those calculations, however, would require numerous assumptions, the selection of which may produce a different curve but with a similar general relationship.

❖ ***Once-through Systems***

For once-through cooling systems, the steam cycle condenser cooling water inlet temperature is also the temperature of the source water. Note that the outlet temperature of the cooling water is typically 15 - 20 °F higher than the inlet temperature. This difference is referred to as the "range." The practical limit of the outlet temperature is approximately 100 °F, since many NPDES permits have limitations in the vicinity of 102 - 105 °F . This does not appear to present a problem, since the maximum monthly average surface water temperature at Jacksonville, Florida (selected by EPA as representing warmer U.S. surface waters) was 83.5 °F which would, using the range values above, result in an effluent temperature of 98.5 - 103.5 °F. To gauge the turbine efficiency energy penalty for once-through cooling systems, the temperature of the source water must be known. These temperatures will vary with location and time of year and estimates for several selected locations are presented in Table 5-7 below.

❖ ***Wet Cooling Towers***

For wet cooling towers, the temperature of the cooling tower outlet is the same as the condenser cooling water inlet temperature. The performance of the cooling tower in terms of the temperature of the cooling tower outlet is a function of the wet bulb temperature of the ambient air and the tower type, size, design, and operation. The wet bulb temperature is a function of the ambient air temperature and the humidity. Wet bulb thermometers were historically used to estimate relative humidity and consist of a standard thermometer with the bulb encircled with a wet piece of cloth. Thus, the temperature read from a wet bulb thermometer includes the cooling effect of water evaporation.

Of all of the tower design parameters, the temperature difference between the wet bulb temperature and the cooling tower outlet (referred to as the "approach") is the most useful in estimating tower performance. The wet cooling tower cooling water outlet temperature of the systems that were used in the analysis for the regulatory options had a design approach of 10 °F. Note that the design approach value is equal to the difference between the tower cooling water outlet temperature and the ambient wet bulb temperature only at the design wet bulb temperature. The actual approach value at wet bulb temperatures other than the design value will vary as described below.

The selection of a 10 °F design approach is based on the data in Attachment C for recently constructed towers. Moreover, a 10 °F approach is considered conservative. As can be seen in Attachment D, a plot of the tower size factor versus the approach shows that a 10 °F approach has a tower size factor of 1.5. The approach is a key factor in sizing towers and has significant cost implications. The trade-off between selecting a small approach versus a higher value is a trade-off between greater capital cost investment versus lower potential energy production. In states where the rates of return on energy investments are fixed (say between 12% and 15%), the higher the capital investment, the higher the return.

For the wet cooling towers used in this analysis, the steam cycle condenser inlet temperature is set equal to the ambient air wet bulb temperature for the location plus the estimated approach value. A design approach value of 10 °F was selected as the common design value for all locations. However, this value is only applicable to instances when the ambient wet bulb temperature is equal to the design wet bulb temperature. In this analysis, the design wet bulb temperature was selected as the 1 percent exceedence value for the specific selected locations.

Attachment E provides a graph showing the relationship between different ambient wet bulb temperatures and the corresponding approach for a "typical" wet tower. The graph shows that as the ambient wet bulb temperature decreases, the approach value increases. The graph in Attachment E was used as the basis for estimating the change in the approach value as the ambient wet bulb temperature changes from the design value for each location. Differences in the location-specific design wet bulb temperature were incorporated by fitting a second order polynomial equation to the data in this graph. The equation was then modified by adjusting the intercept value such that the approach was equal to 10 °F when the wet bulb temperature was equal to the design 1 percent wet bulb temperature for the selected location. The location-specific equations were then used to estimate the condenser inlet temperatures that correspond to the estimated monthly values for wet bulb temperatures at the selected locations.

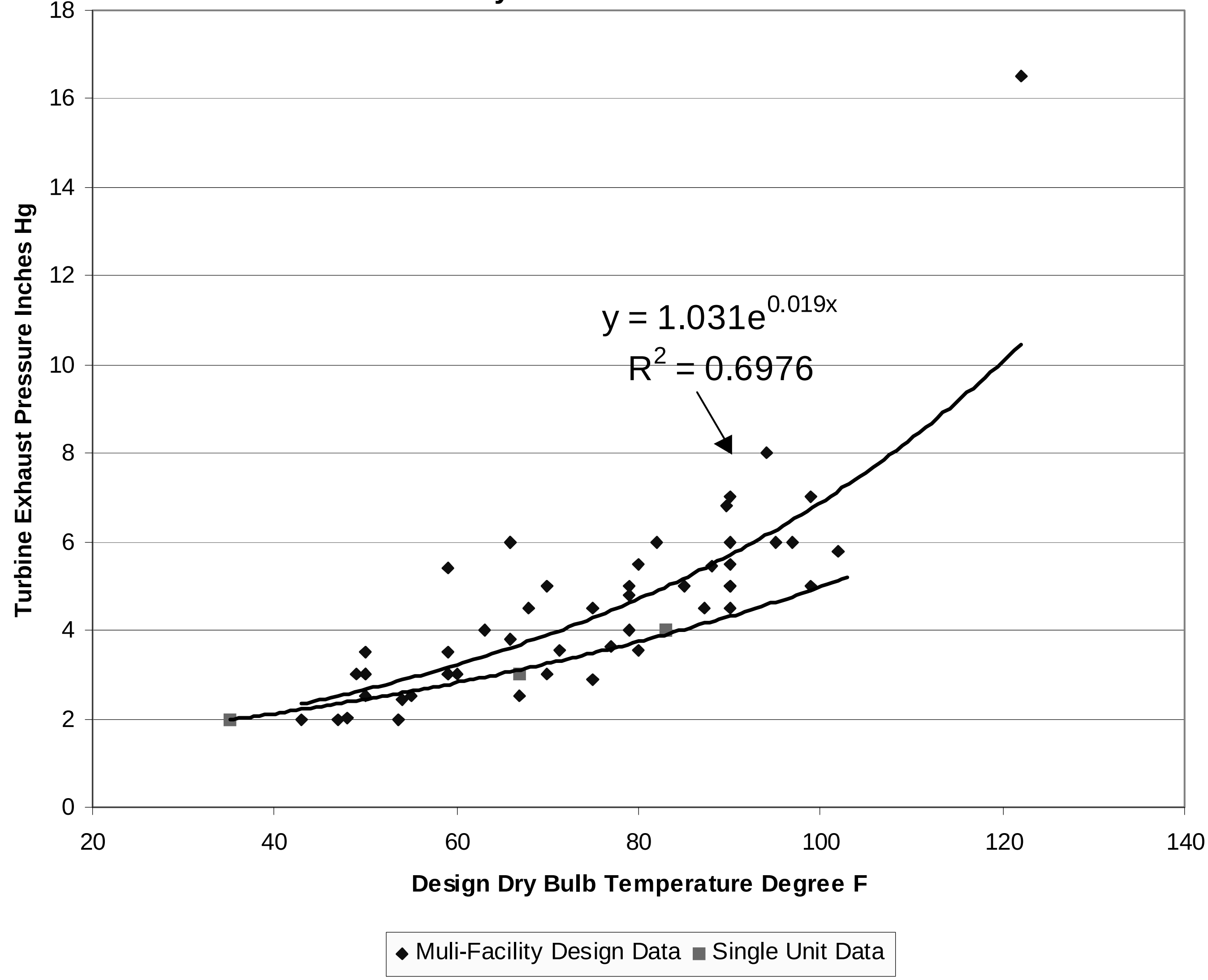
❖ *Air Cooled Condensers*

Air cooled condensers reject heat by conducting it directly from the condensing steam to the ambient air by forcing the air over the heat conducting surface. No evaporation of water is involved. Thus, for air cooled condensers, the condenser performance with regard to turbine exhaust pressure is directly related to the ambient (dry bulb) air temperature, as well as to the condenser design and operating conditions. Note that dry bulb temperature is the same as the standard ambient air temperature with which most people are familiar. Figure 3 presents a plot of the design ambient air temperature and corresponding turbine exhaust pressure for air cooled condensers recently installed by a major cooling system manufacturer (GEA Power Cooling Systems, Inc.). An analysis of the multiple facility data in Figure 3 did not find any trends with respect to plant capacity, location, or age that could justify the separation of these data into subgroups. Three facilities that had very large differences (i.e., >80 ºF) in the design dry bulb temperature compared to the temperature of saturated steam at the exhaust pressure were deleted from the data set used in Figure 3.

A review of the design temperatures indicated that the design temperatures did not always correspond to annual temperature extremes of the location of the plant as might be expected. Thus, it appears that the selection of design values for each application included economic considerations. EPA concluded that these design data represent the range of condenser performance at different temperatures and design conditions. A curve was fitted to the entire set of data to serve as a reasonable means of estimating the relationship of turbine exhaust pressure to different ambient air (dry bulb) temperatures. To validate this approach, condenser performance data for a power plant from an engineering contractor report (Litton, no date) was also plotted. This single plant data produced a flatter curve than the multi-facility plot. In other words, the multi-facility curve predicts a greater increase in turbine exhaust pressure as the dry bulb temperature increases. Therefore, the multi-facility curve was selected as a conservative estimation of the relationship between ambient air temperatures and the turbine exhaust pressure. Note that in the case of air cooled condensers, the turbine exhaust steam pressure includes values above 3.5 inches Hg.

❖
R

Figure 3
Design Dry Bulb and Design Exhaust Pressure for Recently Installed Air Cooled Condensers



egional and Seasonal Data

As noted above, both the source water temperature for once-through cooling systems and the ambient wet bulb and dry bulb temperatures for cooling towers will vary with location and time of year. To estimate average annual energy penalties, EPA sought data to estimate representative monthly values for selected locations. Since plant-specific temperature data may not be available or practical, the conditions for selected locations in different regions are used as examples of the range of possibilities. These four regions include Northeast (Boston, MA), Southeast (Jacksonville, FL), Midwest (Chicago, IL) and Northwest (Seattle, WA). The Southwest Region of the US was not included, since there generally are few once-through systems using surface water in this region.

Table 5-7 presents monthly average coastal water temperatures at the four selected locations. Since the water temperatures remain fairly constant over short periods of time, these data are considered as representative for each month.

Table 5-7: Monthly Average Coastal Water Temperatures (°F)

Location	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Boston, MA[a]	40	36	41	47	56	62	64.5	68	64.5	57	51	42
Jacksonville, FL[a]	57	56	61	69.5	75.5	80.5	83.5	83	82.5	75	67	60
Chicago, IL[b]	39	36	34	36	37	48	61	68	70	63	50	45
Seattle, WA[a]	47	46	46	48.5	50.5	53.5	55.5	56	55.5	53.5	51	49

[a] Source: NOAA Coastal Water Temperature Guides, (www.nodc.noaa.gov/dsdt/cwtg).

[b] Source: Estimate from multi-year plot "Great Lakes Average GLSEA Surface Water Temperature" (http://coastwatch.glerl.noaa.gov/statistics/).

❖ ***Wet and Dry Bulb Temperatures***

Table 5-8 presents design wet bulb temperatures (provided by a cooling system vendor) for the selected locations as the wet bulb temperature that ambient conditions will equal or exceed at selected percent of time (June through September) values. Note that 1 percent represents a period of 29.3 hours. These data, however, represent relatively short periods of time and do not provide any insight as to how the temperatures vary throughout the year. The Agency obtained the *Engineering Weather Data Published by the National Climatic Data Center* to provide monthly wet and dry bulb temperatures. In this data set, wet bulb temperatures were not summarized on a monthly basis, but rather were presented as the average values for different dry bulb temperature ranges along with the average number of hours reported for each range during each month. These hours were further divided into 8-hour periods (midnight to 8AM, 8AM to 4PM, and 4PM to midnight).

Unlike surface water temperature, which tends to change more slowly, the wet bulb and dry bulb temperatures can vary significantly throughout each day and especially from day-to-day. Thus, selecting the temperature to represent the entire month requires some consideration of this variation. The use of daily maximum values would tend to overestimate the overall energy penalty and conversely, the use of 24-hour averages may underestimate the penalty, since the peak power production period is generally during the day.

Since the power demand and ambient wet bulb temperatures tend to peak during the daytime, a time- weighted average of the hourly wet bulb and dry bulb temperatures during the daytime period between 8AM and 4PM was selected as the best method of estimating the ambient wet bulb and dry bulb temperature values to be used in the analysis. The 8AM - 4PM time-weighted average values for wet bulb and dry bulb temperatures were selected as a reasonable compromise between using daily maximum values and 24-hour averages. Table 5-9 presents a summary of the time-weighted wet bulb and dry bulb temperatures for each month for the selected locations. Note that the highest monthly 8AM - 4PM time-weighted average tends to correspond well with the 15 percent exceedence design values. The 15 percent values represent a time period of approximately 18 days which are not necessarily consecutive.

Table 5-8: Design Wet Bulb Temperature Data for Selected Locations

Location	Wet Bulb Temp (°F)			Corresponding Cooling Tower Outlet Temperature (°F)		
	% Time Exceeding			% Time Exceeding		
	1%	5%	15%	1%	5%	15%
Boston, MA	76	73	70	86	83	80
Jacksonville, FL	80	79	77	90	89	87
Chicago, IL	78	75	72	88	85	82
Seattle, WA	66	63	60	76	73	70

Source: www.deltacooling.com

Table 5-9: Time-Weighted Averages for Eight-Hour Period from 8am to 4pm (°F)

Location		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Design 1%
Boston	Wet Bulb	27.5	29.3	36.3	44.6	53.9	62.7	67.9	67.4	61.5	52.0	42.6	32.6	74.0
	Dry Bulb	33.0	35.3	43.2	53.5	63.8	73.9	80.0	78.2	70.4	59.9	49.5	38.4	88.0
Jacksonville	Wet Bulb	52.9	55.3	59.6	64.5	70.3	75.1	77.1	77.1	75.1	69.1	63.1	55.9	79.0
	Dry Bulb	59.8	63.6	70.3	76.6	83.0	87.2	89.3	88.1	85.1	77.8	70.6	62.6	93.0
Chicago	Wet Bulb	23.3	27.0	37.2	46.6	56.6	64.9	69.8	69.3	62.2	51.2	39.1	27.9	76.0
	Dry Bulb	27.6	31.8	43.9	55.7	67.9	77.4	82.5	80.6	72.4	59.9	45.0	32.2	89.0
Seattle	Wet Bulb	39.4	41.8	44.2	47.2	52.0	56.0	59.2	59.6	57.2	51.0	44.0	39.7	65.0
	Dry Bulb	44.3	47.8	51.5	55.6	61.8	67.2	71.6	71.6	67.3	58.1	49.0	44.3	82.0

5.3.4 Calculation of Energy Penalty

Since the energy penalty will vary over time as ambient climatic and source water temperatures vary, the calculation of the total annual energy penalty for a chosen location would best be performed by combining (integrating) the results of individual calculations performed on a periodic basis. For this analysis, a monthly basis was chosen.

The estimated monthly turbine exhaust pressure values for alternative cooling system scenarios were derived using the curves in Figures 2 and 3 in conjunction with the monthly temperature values in Tables 5-7 and 5-9. These turbine exhaust pressure values were then used to estimate the associated change in turbine efficiency using the equations from Figure 1. EPA then calculated the energy penalty for each month. Annual values were calculated by averaging the 12 monthly values.

Tables 5-10 and 5-11 present a summary of the calculated annual average energy penalty values for steam rates of 100 percent and 67 percent of maximum load. These values can be applied directly to the power plant output to determine economic and other impacts. In other words, an energy penalty of 2 percent indicates that the plant output power would be reduced by 2 percent. In addition, Tables 5-10 and 5-11 include the maximum turbine energy penalty associated with maximum design conditions such as once-through systems drawing water at the highest monthly average, and wet towers and air cooled condensers operating in air with a wet bulb and dry bulb temperature at the 1 percent exceedence level. EPA notes that the maximum design values result from using the maximum monthly water temperatures from Table 5-7 and the 1% percent exceedence wet bulb and dry bulb temperatures from Table 5-8.

EPA notes that the penalties presented in Tables 5-10 and 5-11 **do not** comprise the total energy penalties (which incorporate all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 5.1 above. The tables below only present the turbine efficiency penalty. Section 5.4 presents the fan and pumping components of the energy penalty.

Table 5-10: Calculated Energy Penalties for the Turbine Efficiency Component at 100 Pecent of Maximum Steam Load

Location	Cooling Type	Percent Maximum Load	Nuclear Maximum Design	Nuclear Annual Average	Combined Cycle Maximum Design	Combined Cycle Annual Average	Fossil Fuel Maximum Design	Fossil Fuel Annual Average
Boston	Wet Tower vs. Once-through	100%	1.25%	0.37%	0.23%	0.05%	1.09%	0.35%
	Dry Tower vs. Once-through	100%	9.22%	2.85%	2.04%	0.55%	7.76%	2.48%
	Dry Tower vs. Wet Tower	100%	7.96%	2.48%	1.81%	0.50%	6.66%	2.13%
Jacksonville	Wet Tower vs. Once-through	100%	0.71%	0.54%	0.14%	0.10%	0.61%	0.38%
	Dry Tower vs. Once-through	100%	9.86%	6.21%	2.30%	1.35%	8.22%	5.16%
	Dry Tower vs. Wet Tower	100%	9.14%	5.68%	2.16%	1.25%	7.61%	4.78%
Chicago	Wet Tower vs. Once-through	100%	1.39%	0.42%	0.26%	0.05%	1.21%	0.40%
	Dry Tower vs. Once-through	100%	9.47%	3.09%	2.12%	0.60%	7.96%	2.68%
	Dry Tower vs. Wet Tower	100%	8.08%	2.67%	1.85%	0.55%	6.75%	2.28%
Seattle	Wet Tower vs. Once-through	100%	0.77%	0.29%	0.12%	0.03%	0.70%	0.28%
	Dry Tower vs. Once-through	100%	7.60%	2.63%	1.61%	0.49%	6.46%	2.30%
	Dry Tower vs. Wet Tower	100%	6.83%	2.34%	1.48%	0.45%	5.76%	2.02%
Average	Wet Tower vs. Once-through	100%	1.03%	0.40%	0.19%	0.06%	0.90%	0.35%
	Dry Tower vs. Once-through	100%	9.04%	3.70%	2.02%	0.75%	7.60%	3.15%
	Dry Tower vs. Wet Tower	100%	8.00%	3.29%	1.83%	0.69%	6.70%	2.80%

Note: See Section 5.1 for the total energy penalties. This table presents only the turbine component of the total energy penalty.

Table 5-11: Calculated Energy Penalties for the Turbine Efficiency Component at 67% Pecent of Maximum Steam Load

Location	Cooling Type	Percent Maximum Load	Nuclear Maximum Design	Nuclear Annual Average	Combined Cycle Maximum Design	Combined Cycle Annual Average	Fossil Fuel Maximum Design	Fossil Fuel Annual Average
Boston	Wet Tower vs. Once-through	67%	2.32%	0.73%	0.42%	0.14%	2.04%	0.88%
	Dry Tower vs. Once-through	67%	13.82%	4.96%	3.20%	0.98%	15.15%	4.69%
	Dry Tower vs. Wet Tower	67%	11.50%	4.23%	2.78%	0.84%	13.11%	3.81%
Jacksonville	Wet Tower vs. Once-through	67%	1.22%	1.03%	0.24%	0.18%	1.08%	0.93%
	Dry Tower vs. Once-through	67%	13.61%	9.63%	3.50%	2.14%	16.96%	10.06%
	Dry Tower vs. Wet Tower	67%	12.39%	8.60%	3.27%	1.96%	15.88%	9.14%
Chicago	Wet Tower vs. Once-through	67%	2.53%	0.98%	0.47%	0.16%	2.23%	1.02%
	Dry Tower vs. Once-through	67%	14.03%	5.39%	3.30%	1.07%	15.67%	5.30%
	Dry Tower vs. Wet Tower	67%	11.50%	4.41%	2.83%	0.91%	13.44%	4.27%
Seattle	Wet Tower vs. Once-through	67%	1.60%	0.67%	0.27%	0.11%	1.50%	0.74%
	Dry Tower vs. Once-through	67%	12.16%	4.60%	2.60%	0.90%	12.31%	4.50%
	Dry Tower vs. Wet Tower	67%	10.56%	3.93%	2.33%	0.79%	10.81%	3.75%
Average	Wet Tower vs. Once-through	67%	1.92%	0.85%	0.35%	0.15%	1.71%	0.89%
	Dry Tower vs. Once-through	67%	13.41%	6.14%	3.15%	1.27%	15.02%	6.14%
	Dry Tower vs. Wet Tower	67%	11.49%	5.29%	2.80%	1.12%	13.31%	5.24%

Note: See Section 5.1 for the total energy penalties. This table presents only the turbine component of the total energy penalty.

5.4 Energy Penalty Associated with Cooling System Energy Requirements

This analysis is presented to evaluate the energy requirements associated with the operation of the alternative types of cooling systems. As noted previously, the reductions in energy output resulting from the energy required to operate the cooling system equipment are often referred to as parasitic losses. In evaluating this component of the energy penalty, it is the differences between the parasitic losses of the alternative systems that are important. In general, the costs associated with the cooling system energy requirements have been included within the annual O&M cost values for certain regulatory options developed using the methodologies presented in Chapter 2 of this document. Thus, the costs of the cooling system operating energy requirements do not need to be factored into the overall energy penalty cost analysis as a separate value.

Alternative cooling systems can create additional energy demands primarily through the use of fans and pumps. There are other energy demands such as treatment of tower blowdown, but these are insignificant compared to the pump and fan requirements and will not be included here. Some seasonal variation may be expected due to reduced requirements for cooling media flow volume during colder periods. These reduced requirements can include reduced cooling water pumping for once-through systems and reduced fan energy requirements for both wet and dry towers. However, no adjustments were made concerning the potential seasonal variations in cooling water pumping. The seasonal variation in fan power requirements is accounted for in this evaluation by applying an annual fan usage rate. The pumping energy estimates are calculated using a selected cooling water flow rate of 100,000 gpm (223 cfs).

5.4.1 Fan Power Requirements

❖ ***Wet Towers***

In the reference *Cooling Tower Technology* (Burger 1995), several examples are provided for cooling towers with flow rates of 20,000 gpm using 4 cells with either 75 (example #1) or 100 Hp (example #2) fans each. The primary difference between these two examples is that the tower with the higher fan power requirement has an approach of 5 ^{o}F compared to 11 ^{o}F for the tower with the lower fan power requirement. Using an electric motor efficiency of 92 percent and a fan usage factor of 93 percent (Fleming 2001), the resulting fan electric power requirements are equal to 0.236 MW and 0.314 MW for the four cells with 75 and 100 Hp fan motors, respectively. These example towers both had a heat load of 150 million BTU/hr. Table 5-14 provides the percent of power output penalty based on equivalent plant capacities derived using the heat rejection factors described below. Note that fan gear efficiency values are not applicable because they do not affect the fan motor power rating or the amount of electricity required to operate the fan motors.

A third example was provided in vendor-supplied data (Fleming 2001), in which a cooling tower with a cooling water flow rate of 243,000 gpm had a total fan motor capacity brake-Hp of 250 for each of 12 cells. This wet tower had a design temperature range of 15 °F and an approach of 10 °F. The percent of power output turbine penalty shown in Tables 5-10 and 5-11 is also based on equivalent plant capacities derived using the heat rejection factors described below.

A fourth example is a cross-flow cooling tower for a 35 MW coal-fired plant in Iowa (Litton, no date). In this example, the wet tower consists of two cells with one 150 Hp fan each, with a cooling water flow rate of 30,000 gpm. This wet tower had a design temperature range of 16 °F, an approach of 12 °F, and wet bulb temperature of 78 °F. The calculated energy penalty in this example is 0.67 percent.

Example #2, which has the smallest approach value, represents the high end of the range of calculated wet tower fan energy penalties presented in Table 5-12. Note that smaller approach values correspond to larger, more expensive (both in capital and O&M costs) towers. Since the fossil fuel plant penalty value for example #4, which is based mostly on empirical data, is just below the fossil fuel penalty calculated for example #2, EPA has chosen the calculated values for example #2 as representing a conservative estimate for the wet tower fan energy penalty.

EPA notes that the penalties presented in Tables 5-12 **do not** comprise the total energy penalty (which incorporates all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 3.1 above. The table below only presents the fan component of the penalty.

Table 5-12: Wet Tower Fan Power Energy Penalty

Example Plant	**Range/ Approach** (Degree F)	**Flow** (gpm)	**Fan Power Rating** (Hp)	**Fan Power Required** (MW)	**Plant Type**	**Plant Capacity** (MW)	**Percent of Output** (%)
#1	15/11	20,000	300	0.236	Nuclear	35	0.68%
					Fossil Fuel	43	0.55%
					Comb. Cycle	130	0.18%
#2	15/5	20,000	400	0.314	Nuclear	35	0.91%
					Fossil Fuel	43	0.73%
					Comb. Cycle	130	0.24%
#3	15/10	243,000	3,000	2.357	Nuclear	420	0.56%
					Fossil Fuel	525	0.45%
					Comb. Cycle	1574	0.15%
#4	16/12	30,000	300.0	0.236	Fossil Fuel	35	0.67%

Note: See Section 5.1 for the total energy penalties. This table presents only the fan component of the total energy penalty.

❖ ***Air Cooled Condensers***

Air cooled condensers require greater air flow than recirculating wet towers because they cannot rely on evaporative heat transfer. The fan power requirements are generally greater than those needed by wet towers by a factor of 3 to 4 (Tallon 2001). While the fan power requirements can be substantial, at least a portion of this increase over wet cooling systems is offset by the elimination of the pumping energy requirements associated with wet cooling systems described below.

The El Dorado power plant in Boulder, Nevada which was visited by EPA is a combined-cycle plant that uses air cooled condensers due to the lack of sufficient water resources. This facility is located in a relatively hot section of the U.S. Because the plant has a relatively low design temperature (67 ^{o}F) in a hot environment, it should be considered as representative of a conservative situation with respect to the energy requirements for operating fans in air cooled condensers. The steam portion of the plant has a capacity of 150 MW (1.1 million lb/hr steam flow). The air cooled condensers consist of 30 cells with a 200 Hp fan each. A fan motor efficiency of 92 percent is assumed. Each fan has two operating speeds, with the low speed consuming 20 percent of the fan motor power rating.

The facility manager provided estimates of the proportion of time that the fans were operated at low or full speed during different portions of the year (Tatar 2001). Factoring in the time proportions and the corresponding power requirements results in an overall annual fan power factor of 72 percent for this facility. In other words, over a one year period, the fan power requirement will average 75 percent of the fan motor power rating. A comparison of the climatic data for Las Vegas (located nearby) and Jacksonville, Florida shows that the Jacksonville mean maximum temperature values were slightly warmer in the winter and slightly cooler in the summer. Adjustments in the annual fan power factor calculations to address Jacksonville's slightly warmer winter months resulted in a projected annual fan power factor of 77 percent. EPA chose a factor of 75 percent as representative of warmer regions of the U.S. Due to lack of available operational data for other locations, this value is used for facilities throughout the U.S. and represents an conservative value for the much cooler regions.

Prior to applying this factor, the resulting maximum energy penalty during warmer months is 3.2 percent for the steam portion only. This value is the maximum instantaneous penalty that would be experienced during high temperature conditions. When the annual fan power factor of 75 percent is applied, the annual fan energy penalty becomes 2.4 percent of the plant power output. An engineer from an air cooled condenser manufacturer indicated that the majority of air cooled condensers being installed today also include two-speed fans and that the 20 percent power ratio for the low speed was the factor that they used also. In fact, some dry cooling systems, particularly those in very cold regions, use fans with variable speed drives to provide even better operational control. Similar calculations for a waste-to-energy plant in Spokane, Washington resulted in a maximum fan operating penalty of 2.8 percent and an annual average of 2.1 percent using the 75 percent fan power factor. Thus, the factor of 2.4 percent selected by EPA as a conservative annual penalty value appears valid.

5.4.2 Cooling Water Pumping Requirements

The Agency notes that it conducted the following analysis for new, "greenfield" facilities and transferred the results of this analysis to the cooling system conversions for existing facilities considered as regulatory options for this proposal. As discussed in Section 5.6 below, the Department of Energy (DOE) concludes in their draft energy penalty analysis that the pumping component of the energy penalty for existing facilities may be higher than calculated herein by EPA for new, "greenfield" facilities.

The energy requirements for cooling water pumping can be estimated by combining the flow rates and the total head (usually given in feet of water) that must be pumped. Estimating the power requirements for the alternative cooling systems that use water is somewhat complex in that there are several components to the total pumping head involved. For example, a once-through system must pump water from the water source to the steam condensers, which will include both a static head from the elevation of the source to the condenser (use of groundwater would represent an extreme case) and friction head losses through the piping and the condenser. The pipe friction head is dependent on the distance between the power plant and the source plus the size and number of pipes, pipe fittings, and the flow rate. The condenser friction head loss is a function of the condenser design and flow rate.

Wet cooling towers must also pump water against both a static and friction head. A power plant engineering consultant estimated that the total pumping head at a typical once-through facility would be approximately 50 ft (Taylor 2001). EPA performed a detailed analysis of the cooling water pumping head that would result from different combinations of piping velocities and distances. The results of this analysis showed that the pumping head was in many scenarios similar in value for both once-through and wet towers, and that the estimated pumping head ranged from approximately 40 to 60 feet depending on the assumed values. Since EPA's analysis produced similar values as the 50 ft pumping head provided by the engineering consultant, this value was used in the estimation of the pumping requirements for cooling water intakes for both once-through and wet tower systems. The following sections describe the method for deriving these pumping head values.

❖ *Friction Losses*

In order to provide a point of comparison, a cooling water flow rate of 100,000 gpm (223 cfs) was used. A recently reported general pipe sizing rule indicating that a pipe flow velocity of 5.7 fps is the optimum flow rate with regards to the competing cost values was used as the starting point for flow velocity (Durand et al. 1999). Such a minimum velocity is needed to prevent sediment deposition and pipe fouling. Using this criterion as a starting point, four 42-inch steel pipes carrying 25,000 gpm each at a velocity of 5.8 fps were selected. Each pipe would have a friction head loss of 0.358 ft/100 ft of pipe (Permutit 1961), resulting in a friction loss of 3.6 ft for every 1,000 ft of length. Since capital costs may dictate using fewer pipes with greater pipe flow rates, two other scenarios using either three or two parallel 42-inch pipes were also evaluated. Three pipes would result in a flow rate and velocity of 33,000 gpm and 7.7 fps, which results in a friction head loss of 6.1 ft/1000ft. Two pipes would result in a flow rate and velocity of 50,000 gpm and 11.6 fps, which results in a friction head loss of 12.8 ft/1000ft. The estimated 50 ft total pumping head was most consistent with a pipe velocity of 7.7 fps (three 42-inch pipes).

The relative distances of the power plant condensers to the once-through cooling water intakes as compared to the distance from the plant to the alternative cooling tower can be an important factor. In general, the distances that the large volumes of cooling water must be pumped will be greater for once-through cooling systems. For this analysis, a fixed distance of 300 ft was selected for the cooling tower. Various distances ranging from 300 ft to 3,000 ft are used for the once-through system. The friction head was also assumed to include miscellaneous losses due to inlets, outlets, bends, valves, etc., which can be calculated using equivalent lengths of pipe. For 42-in. steel pipe, each entrance and long sweep elbow is equal to about 60 ft in added pipe length. For the purposes of this analysis, both systems were assumed to have five such fittings for an added length of 300 ft. The engineering estimate of 50 ft for pumping head was most consistent with a once-through pumping distance of approximately 1,000 ft.

❖ *Static Head*

Static head refers to the distance in height that the water must be pumped from the source elevation to the destination. In the case of once-through cooling systems, this is the distance in elevation between the source water and the condenser inlet. However, many power plants eliminate a significant portion of the static head loss by operating the condenser piping as a siphon. This is done by installing vacuum pumps at the high point of the water loop. In EPA's analysis, a static head of 20 ft produced a total pumping head value that was most consistent with the engineering consultant's estimate of 50 feet.

In the case of cooling towers, static head is related to the height of the tower, and vendor data for the overall pumping head through the tower is available. This pumping head includes both the static and dynamic heads within the tower, but was included as the static head component for the analysis. Vendor data reported a total pumping head of 25 ft for a large cooling tower sized to handle 335,000 gpm (Fleming 2001). The tower is a counter-flow packed tower design. Adding the condenser losses and pipe losses resulted in a total pumping head of approximately 50 feet.

❖ ***Condenser Losses***

Condenser design data provided by a condenser manufacturer, Graham Corporation, showed condenser head losses ranging from 21 ft of water for small condensers (cooling flow <50,000 gpm) to 41 ft for larger condensers (Hess 2001). Another source showed head losses through the tubes of a large condenser (311,000 gpm) to be approximately 9 ft of water (HES. 2001). For the purposes of this analysis, EPA estimated condenser head losses to be 20 ft of water. For comparable systems with similar cooling water flow rates, the condenser head loss component should be the same for both once-through systems and recirculating wet towers.

❖ ***Flow Rates***

In general, the cooling water flow rate is a function of the heat rejection rate through the condensers and the range of temperature between the condenser inlet and outlet. The flow rate for cooling towers is approximately 95 percent that of once-through cooling water systems, depending on the cooling temperature range. However, cooling tower systems also still require some pumping of make-up water. For the purposes of this analysis, the flow rates for each system will be assumed to be essentially the same. All values used in the calculations are for a cooling water flow rate of 100,000 gpm. Values for larger and smaller systems can be factored against these values. The total pump and motor efficiency is assumed to be equal to 70 percent.

5.5 Analysis of Cooling System Energy Requirements

This analysis evaluates the energy penalty associated with the operation of cooling system equipment for conversion from once-through systems to wet towers and for conversion to air cooled systems by estimating the net difference in required pumping and fan energy between the systems. This penalty can then be compared to the power output associated with a cooling flow rate of 100,000 gpm to derive a percent of plant output figure that is a similar measure to the turbine efficiency penalty described earlier. The power output was determined by comparing condenser heat rejection rates for different types of systems. As noted earlier, the cost of this energy penalty component has already been included in the alternative cooling system O&M costs discussed in Chapter 2 of this document, but was derived independently for this analysis.

Table 5-13 shows the pumping head and energy requirements for pumping 100,000 gpm of cooling water for both once-through and recirculating wet towers using the various piping scenario assumptions. In general, the comparison of two types of cooling systems shows offsetting energy requirements that essentially show zero pumping penalty between once-through and wet towers as the pumping distance for the once-through system increases to approximately 1,000 ft. In fact, it is apparent that for once-through systems with higher pipe velocities and pumping distances, more cooling water pumping energy may be required for the once-through system than for a wet cooling tower. Thus, when converting from once-through to recirculating wet towers, the differences in pumping energy requirements may be relatively small.

As described above, wet towers will require additional energy to operate the fans, which results in a net increase in the energy needed to operate the wet tower cooling system compared to once-through. Note that the average calculated pumping head across the various scenarios for once-through systems was 54 ft. This data suggests that an average pumping head of 50 feet for once-through systems appears to be a reasonable assumption where specific data are not available.

EPA notes that the penalties presented in Tables 5-13 and 5-14 **do not** comprise the total energy penalties (which incorporate all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 3.1 above. The tables below only present the pumping components.

Table 5-13: Cooling Water Pumping Head and Energy for 100,000 gpm System Wet Towers Versus Once-through At 20' Static Head

Cooling System Type	Distance Pumped	Static Head	Condenser Head	Equiv. Length Misc. Losses	Pipe Velocity	Friction Loss Rate	Friction Head	Total Head	Net Difference	Flow Rate	Hydraulic-Hp	Brake-Hp	Power Required	Energy Penalty
	ft.	ft.	ft	ft.	fps	ft/1,000ft	ft.	ft.	ft	gpm	Hp	Hp	kW	kW
Once-through at 20' Static Head Using 4: 42" Pipes at 300' Length														
Once-through	300	20	21	300	5.8	3.6	2	43		100,000	1089	1556	1161	
Wet Tower	300	25	21	300	5.8	3.6	2	48	5	100,000	1216	1737	1296	135
Once-through at 20' Static Head Using 3: 42" Pipes at 300' Length														
Once-through	300	20	21	300	7.7	6.1	4	45		100,000	1127	1610	1201	
Wet Tower	300	25	21	300	7.7	6.1	4	50	5	100,000	1254	1791	1336	135
Once-through at 20' Static Head Using 2: 42" Pipes at 300' Length														
Once-through	300	20	21	300	11.6	12.8	8	49		100,000	1229	1755	1310	
Wet Tower	300	25	21	300	11.6	12.8	8	54	5	100,000	1355	1936	1444	135
Once-through at 20' Static Head Using 4: 42" Pipes at 1000' Length														
Once-through	1000	20	21	300	5.8	3.6	5	46		100,000	1153	1647	1229	
Wet Tower	300	25	21	300	5.8	3.6	2	48	2	100,000	1216	1737	1296	67
Once-through at 20' Static Head Using 3: 42" Pipes at 1000' Length														
Once-through	1000	20	21	300	7.7	6.1	8	49		100,000	1235	1764	1316	
Wet Tower	300	25	21	300	7.7	6.1	4	50	1	100,000	1254	1791	1336	20
Once-through at 20' Static Head Using 2: 42" Pipes at 1000' Length														
Once-through	1000	20	21	300	11.6	12.8	17	58		100,000	1455	2079	1551	
Wet Tower	300	25	21	300	11.6	12.8	8	54	-4	100,000	1355	1936	1444	-107
Once-through at 20' Static Head Using 4: 42" Pipes at 3000' Length														
Once-through	3000	20	21	300	5.8	3.6	12	53		100,000	1335	1907	1423	
Wet Tower	300	25	21	300	5.8	3.6	2	48	-5	100,000	1216	1737	1296	-127
Once-through at 20' Static Head Using 3: 42" Pipes at 3000' Length														
Once-through	3000	20	21	300	7.7	6.1	20	61		100,000	1543	2204	1644	
Wet Tower	300	25	21	300	7.7	6.1	4	50	-11	100,000	1254	1791	1336	-309
Once-through at 20' Static Head Using 2: 42" Pipes at 3000' Length														
Once-through	3000	20	21	300	11.6	12.8	42	83		100,000	2101	3002	2239	
Wet Tower	300	25	21	300	11.6	12.8	8	54	-30	100,000	1355	1936	1444	-795

Note: Wet Towers are assumed to always be at 300' distance and have the same tower pumping head of 25' in all scenarios shown.

The same flow rate of 100,000gpm (223 cfs) is used for all scenarios.

See Section 5.1 for the total energy penalties. This table presents only the pumping component of the total energy penalty.

❖ *Cooling System Energy Requirements Penalty as Percent of Power Output*

One method of estimating the capacity of a power plant associated with a given cooling flow rate is to compute the heat rejected by the cooling system and determine the capacity that would match this rejection rate for a "typical" power plant in each category. In order to determine the cooling system heat rejection rate, both the cooling flow (100,000 gpm) and the condenser temperature range between inlet and outlet must be estimated. In addition, the capacity that corresponds to the power plant heat rejection rate must be determined. The heat rejection rate is directly related to the type, design, and capacity of a power plant. The method used here was to determine the ratio of the plant capacity divided by the heat rejection rate as measured in equivalent electric power.

An analysis of condenser cooling water flow rates, temperature ranges and power outputs for several existing nuclear plants provided ratios of the plant output to the power equivalent of heat rejection ranging from 0.75 to 0.92. A similar analysis for coal-fired power plants provided ratios ranging from 1.0 to 1.45. Use of a lower factor results in a lower power plant capacity estimate and, consequently, a higher value for the energy requirement as a percent of capacity. Therefore, EPA chose to use values near the lower end of the range observed. EPA selected ratios of 0.8 and 1.0 for nuclear and fossil-fueled plants, respectively. The steam portion of a combined cycle plant is assumed to have a factor similar to fossil fuel plants of 1.0. Considering that this applies to only one-third of the total plant output, the overall factor for combined-cycle plants is estimated to be 3.0.

In order to correlate the cooling flow energy requirement data to the power output, a condenser temperature range must also be estimated. A review of data from newly constructed plants in Attachment C showed no immediately discernable pattern on a regional basis for approach or range values. Therefore, these values will not be differentiated on a regional basis in this analysis. The data did, however, indicate a median approach of 10 °F (average 10.4 °F) and a median range of 20 °F (average 21.1 °F). This range value is consistent with the value assumed in other EPA analyses and therefore a range of 20 °F will be used. Table 5-14 presents the energy penalties corresponding to the pumping energy requirements from Table 5-13 using the above factors.

Table 5-14: Comparison of Pumping Power Requirement and Energy Penalty to Power Plant Output

Cooling system Type	Distance Pumped	Static Head	Power Required	Flow Rate	Range	Nuclear Power/ Heat	Nuclear Equiv. Output	Nuclear Pumping	Fossil Fuel Power/ Heat	Fossil Fuel Equiv. Output	Fossil Fuel Pumping	Comb.-Cycle Power/ Heat	Comb.-Cycle Equiv.	Comb.-Cycle Pumping
	ft.	ft.	kW	gpm	ºF	Ratio	(MW)	% of Output	Ratio	(MW)	% of Output	Ratio	Output (MW)	% of Output
Once-through at 20' Static Head Using 4: 42" Pipes at 300' Length														
Once-through	300	20	1161.1	100,000	20	0.8	235	0.49%	1	294	0.39%	3	882	0.13%
Wet Tower	300	25	1295.6	100,000	20	0.8	235	0.55%	1	294	0.44%	3	882	0.15%
Once-through at 20' Static Head Using 3: 42" Pipes at 300' Length														
Once-through	300	20	1201.4	100,000	20	0.8	235	0.51%	1	294	0.41%	3	882	0.14%
Wet Tower	300	25	1335.9	100,000	20	0.8	235	0.57%	1	294	0.45%	3	882	0.15%
Once-through at 20' Static Head Using 2: 42" Pipes at 300' Length														
Once-through	300	20	1309.6	100,000	20	0.8	235	0.56%	1	294	0.45%	3	882	0.15%
Wet Tower	300	25	1444.1	100,000	20	0.8	235	0.61%	1	294	0.49%	3	882	0.16%
Once-through at 20' Static Head Using 4: 42" Pipes at 1000' Length														
Once-through	1000	20	1228.8	100,000	20	0.8	235	0.52%	1	294	0.42%	3	882	0.14%
Wet Tower	300	25	1295.6	100,000	20	0.8	235	0.55%	1	294	0.44%	3	882	0.15%
Once-through at 20' Static Head Using 3: 42" Pipes at 1000' Length														
Once-through	1000	20	1316.3	100,000	20	0.8	235	0.56%	1	294	0.45%	3	882	0.15%
Wet Tower	300	25	1335.9	100,000	20	0.8	235	0.57%	1	294	0.45%	3	882	0.15%
Once-through at 20' Static Head Using 2: 42" Pipes at 1000' Length														
Once-through	1000	20	1550.6	100,000	20	0.8	235	0.66%	1	294	0.53%	3	882	0.18%
Wet Tower	300	25	1444.1	100,000	20	0.8	235	0.61%	1	294	0.49%	3	882	0.16%
Once-through at 20' Static Head Using 4: 42" Pipes at 3000' Length														
Once-through	3000	20	1422.5	100,000	20	0.8	235	0.60%	1	294	0.48%	3	882	0.16%
Wet Tower	300	25	1295.6	100,000	20	0.8	235	0.55%	1	294	0.44%	3	882	0.15%
Once-through at 20' Static Head Using 3: 42" Pipes at 3000' Length														
Once-through	3000	20	1644.5	100,000	20	0.8	235	0.70%	1	294	0.56%	3	882	0.19%
Wet Tower	300	25	1335.9	100,000	20	0.8	235	0.57%	1	294	0.45%	3	882	0.15%
Once-through at 20' Static Head Using 2: 42" Pipes at 3000' Length														
Once-through	3000	20	2239.3	100,000	20	0.8	235	0.95%	1	294	0.76%	3	882	0.25%
Wet Tower	300	25	1444.1	100,000	20	0.8	235	0.61%	1	294	0.49%	3	882	0.16%

Note: Wet Towers are assumed to always be at 300' distance and have the same tower pumping head of 25' in all scenarios shown. The same flow rate cfs) is used for all scenarios. Power/Heat Ratio refers to the ratio of Power Plant Output (MW) to the heat (in equivalent MW) transferred through the 3-1 for the total energy penalties. This table presents only the pumping component of the total energy penalty

5.5.1 Summary of Cooling System Energy Requirements

EPA chose the piping scenario in Table 5-13 where pumping head is close to 50 ft for both once-through and recirculating systems at new, "greenfield" facilities (that is, once-through at 1,000 ft and 3-42 in. pipes in Table 5-13). Thus, the cooling water pumping requirements for once-through and recirculating wet towers are nearly equal using the chosen site-specific conditions. Table 5-15 summarizes the fan and pumping equipment energy requirements as a percent of power output for each type of power plant. Table 5-16 presents the net difference in energy requirements shown in Table 5-15 for the alternative cooling systems. The net differences in Table 5-16 are the equipment operating energy penalties associated with conversion from one cooling technology to another.

EPA notes that the penalties presented in Tables 5-15 and 5-16 **do not** comprise the total energy penalties (which incorporate all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 5.1 above. The tables below only present the pumping and fan components. Section 5.3.4 presents the turbine efficiency components of the energy penalty.

Table 5-15: Summary of Fan and Pumping Energy Requirements as a Percent of Power Output

	Wet Tower Pumping	Wet Tower Fan	Wet Tower Total	Once-through Total (Pumping)	Dry Tower Total (Fan)
Nuclear	0.57%	0.91%	1.48%	0.56%	3.04%
Fossil Fuel	0.45%	0.73%	1.18%	0.45%	2.43%
Combined-Cycle	0.15%	0.24%	0.39%	0.15%	0.81%

Note: See Section 5.1 for the total energy penalties.

Table 5-16: Fan and Pumping Energy Penalty Associated with Alternative Cooling System as a Percent of Power Output

	Wet Tower Vs Once-through	Dry Tower Vs Wet Tower	Dry Tower Vs Once-through
Nuclear	0.92%	1.56%	2.48%
Fossil Fuel	0.73%	1.25%	1.98%
Combined-Cycle	0.24%	0.42%	0.66%

Note: See Section 5.1 for the total energy penalties.

5.6 Other Sources of Energy Penalty Estimates

The Agency sought out additional sources of energy penalty estimates for its analysis of regulatory options for the 316(b) Existing Facility proposal. In part due to the lack of robust, empirical data available, the Agency undertook the original energy penalty analysis in support of the New Facility Rule. For this Existing Facility proposal the fact that certain regulatory options involved the conversion of aging cooling systems at existing facilities presented an additional complexity to the Agency. The following sections summarize the Agency's data collection for estimates of energy penalties at existing facilities.

5.6.1 Jefferies Generating Station Energy Penalty Study

As a result of its research for empirical examples of cooling system conversions, the Agency identified an empirical energy penalty study associated with the construction of wet, mechanical-draft cooling towers to replace an original once-through system. The Jefferies Generating Station -- a 346 MW, coal-fired plant in South Carolina – owned by Santee Cooper, conducted a turbine efficiency loss study in the late 1980s. The facility converted their cooling system (after many years of operation utilizing a once-through system) to a full recirculating, mechanical-draft system around 1985. Due to the unusual arrangement whereby the U.S. Army Corps of Engineers (USACE) paid for the construction and operation of the cooling tower, Santee Cooper began an empirical study to assess the economic impact of the operation of the cooling towers over the previous once-through system, in order to obtain reimbursement from the USACE. The study lasted several years (1985 to 1990). However, the empirical stage of data gathering occurred primarily in 1988. Santee Cooper determined (and the USACE eventually agreed) that the cooling tower had decreased the efficiency of each of the plant's steam turbines. The efficiency penalties determined by Santee Cooper were a maximum of 0.97 percent of plant capacity (for both units, combined) and an annual average of 0.16 percent for the year 1988. Note, that because the USACE maintains and pays for the operation of the cooling towers, Santee Cooper only examined the turbine portion of the energy penalty at the plant. The Agency requested documentation on the historic operation of the towers from the USACE (in addition to the construction costs from 1986) but did not receive this information at the time of publication of this proposal. The study conducted by Santee Cooper is included in the record of today's proposal (see DCN 4-2527). The Agency notes that its fossil-fuel estimate for the national-average, peak-summer, turbine energy penalty is 0.90 percent and the mean-annual, national-average energy penalty is 0.35 percent (at 100 percent of maximum load). For the model plant in Jacksonville, Florida the Agency calculated a fossil-fuel peak-summer turbine energy penalty of 0.61 percent and the mean-annual turbine energy penalty of 0.38 percent (at 100 percent of maximum load).

5.6.2 U.S. Department of Energy Peak-Summer Energy Penalty Study

The U.S. Department of Energy (DOE), through its Office of Fossil Energy, National Energy Technology Laboratory (NETL), and Argonne National Laboratory (ANL), studied the energy penalty resulting from converting plants with once-through cooling to wet towers or indirect dry towers. DOE modeled five locations – Delaware River Basin (Philadelphia), Michigan/Great Lakes (Detroit), Ohio River Valley (Indianapolis), South (Atlanta), and Southwest (Yuma) – using an ASPEN simulator model. The model evaluated the performance and energy penalty for hypothetical 400-MW coal-fired plants that were retrofitted from using once-through cooling systems to indirect-wet- and indirect-dry-recirculating systems. The modeling was done to simulate the hottest time of the year using temperature input values that are exceeded only 1 percent of the time between June through September at each modeled location. At DOE's request, EPA provided, discharge temperature data and thermal discharge permit limits for facilities at or near the DOE study locations for use in the model. EPA also provided comments regarding the framework of the modeling project, which are included in the record of this proposal (see DCN 4-2512)

After completing their initial modeling, DOE shared the results of their working draft report with the EPA, which is included in the record of this proposal (see DCN 4-2511). DOE estimates that conversion to a wet tower could cause peak-summer energy penalties ranging from 2.8 percent to 4.0 percent. Therefore, DOE estimates that the plant will produce 2.8 percent to 4.0 percent less electricity with a wet tower than it did with a once-through system while burning the same amount of coal. Further, DOE estimates that conversion to an indirect-dry tower could cause peak-summer energy penalties ranging from 8.9 percent to 14.1 percent with a design approach of 20 degrees Fahrenheit and 12.7 percent to approximately 18 percent with an approach of 40 degrees Fahrenheit.

EPA did not model indirect-dry cooling systems, and therefore cannot directly compare its estimates to those developed by DOE. However, EPA can compare its estimates of peak summer energy penalties for mechanical draft wet cooling towers to those developed by DOE. The Agency finds that its estimates of peak summer energy penalties are significantly lower than those developed by DOE (see section 5.1 for EPA's estimates of peak summer energy penalties of mechanical draft cooling towers). EPA and DOE believe that the difference in these estimates is most likely due to two key factors: (1) the estimated energy penalty attributable to the parasitic energy use of cooling water pumps and (2) the estimated design temperature ranges of cooling water from condenser inlet to outlet.

As discussed at Section 5.3 above, EPA developed energy penalty estimates for this proposal based on its estimates for the 316(b) New Facility Rule. For the energy penalty estimates of the 316(b) New Facility Rule, the Agency conducted an analysis of a variety of pumping scenarios for once-through versus recirculating systems at new "greenfield" facilities. The Agency concluded that for "greenfield" facilities, the cooling towers would generally be sited in close proximity to condenser units. Therefore, the Agency estimated that pumping distances for recirculating systems would be significantly less than those for once-through systems. (The Agency provided this analysis for public comment in the June 2001 Notice of Data Availability). In the analysis of energy penalties for new, "greenfield" plants, the Agency concluded that the difference in pumping distance for a once-through system would offset the additional static head pumping requirements of a typical mechanical draft cooling tower (see section 5.4.2 for the Agency's analysis of pumping energy requirements). Therefore, the analysis of energy penalties used by the Agency for this proposal estimates 0.0 percent energy penalty due to pumping requirements. DOE, on the other hand, estimates that a retrofitted wet cooling tower would require significantly more pumping energy than a baseline, once-through cooling system. The results of the DOE study show pumping energy penalties ranging from 0.2 to 0.7 percent. The Agency views the DOE estimates to be reasonable for a variety of retrofit scenarios at existing facilities and will reconsider this subject in the analysis of regulatory options for the final rule.

While EPA did not directly estimate design temperature range in its modeling approach, in effect DOE and EPA used different design temperature ranges, which can dramatically affect energy penalty estimates. The DOE modeling approach used simulated inlet and discharge water temperatures for the chosen sites. EPA provided thermal discharge permit information, which DOE incorporated into a parametric analysis of design temperature ranges (that is, DOE examined a variety of temperature ranges, from 5 degrees F to 25 degrees F). For example, the design ranges examined by DOE for their Michigan site show peak energy penalties that vary by 1 percent, from 3.95 % for a 7 degree F design range to 2.94 % for a 25 degree F design range. In the case of other model sites, such as Georgia, a design range increase of 5 degrees F (from 5 degrees F to 10 degrees F) can dramatically effect the results of the energy penalty estimates. The DOE model estimates a 3.99 % percent energy penalty for the 5 degrees F design range in Georgia and 2.78 % for the 10 degrees F design range assumption. As EPA noted in its comments on DOE's proposed energy penalty analysis, EPA believes that design temperature ranges of less than 13 degrees F are not realistic at most locations and are likely to lead to energy penalty estimates that are higher than would occur under realistic operating conditions.

In addition to the two key factors described above, DOE expressed concern to EPA that the Agency's modeling analysis of turbine energy penalties did not incorporate subtle effects on the condenser duty. Specifically, DOE did not believe that the Agency's model takes into account the increase in turbine exhaust temperature (or steam temperature to condenser) resulting from a corresponding increase in condenser duty when changing the once-through cooling water system to a wet cooling tower. Under peak energy penalty periods, the temperature of the condenser cooling water will be greater under wet cooling tower operation then the same plant operated under once-through cooling because of the difference between ambient wet-bulb and surface water temperatures. DOE believes that the increased condenser duty for the wet cooling tower results in an increase in cooling water flow which increases the cooling water pump and cooling tower fan energy penalties compared to the Agency's approach.

DOE also points out that the Agency's model does not consider a second effect that since the steam is condensed at a slightly higher temperature for the wet cooling tower case, the reheating of the recirculated steam condensate will require a reduction in the amount of steam bleed from the turbine system. This results in a slightly higher steam flowrate through the turbines and into the condenser. This again increases the condenser duty and would again increase the parasitic energy penalties. However, this would probably be offset by an increase in power due to a small increase in the steam flowrate in the turbines. DOE estimates that these effects may contribute a maximum of 0.5 percentage points to the Agency 's evaluation of the peak-summer energy penalty.

5.6.3 Catawba and McGuire Nuclear Plant Comparison

One literature source the Agency encountered calculated the energy penalty of a nuclear plant employing a mechanical-draft wet cooling system by comparing the electrical ratings of the Catawba and McGuire Nuclear Plants. Because the two plants were constructed nearly identically, the author hypothesized that the percent difference electrical rating between the two plants would represent the energy usage of a cooling tower. The Agency notes that even though a comparison of this type would theoretically calculate the net energy use of the pumps and fans of the wet cooling tower system as compared to the once-through system, there are a variety of complicating factors that are not accounted for or are overlooked in this case. The electrical rating of a nuclear plant does not, to the Agency's knowledge, account for the turbine efficiency penalty component. This key portion of the energy penalty would not be included in the electrical rating calculations of the plant. The comparison could, therefore, underestimate the total energy penalty of the cooling system.

Nonetheless, the Agency examined the historical energy penalty estimate for the Catawba versus McGuire case and determined that the source had made an error in calculating an estimate of 3 percent for the overall energy use of the cooling towers over the once-through system. The error made was to assume that each plant had the same gross capacity. In fact, the McGuire plant has a gross capacity that is 31 MW greater than Catawba. Therefore, a comparison of the percentage difference between gross and net capacity for the two plants actually should be calculated as 1.7 percent. This energy penalty estimate for the fan and pumping components is higher than that estimated by the Agency elsewhere in this chapte for nuclear facilities. The Agency estimates that the total of the fanning and pumping components for a nuclear plant would be 0.9 percent. As described in Section 5.3 of this chapter, the Agency's estimates of the pumping components developed for new, "greenfield" facilities calculate no net change in the pumping requirements between once-through and recirculating wet cooling tower systems. As stated in Section 5.6.2 above, this may also explain to some degree the differences between the Agency's and the Catawba/McGuire estimate.

5.6.3 Palisades Cooling System Conversion Energy Penalty Estimate

The Agency learned from discussions with, and information submitted by, Consumers Energy that the cooling tower system at Palisades might have a significant impact on the efficiency of the plant's generating unit. Though the plant was unable to provide historical studies of the energy penalty of the cooling tower system, they estimate that the effect could be approximately 7 percent (Gulvas, 2002). Prior to the 1970's conversion, the Nuclear Regulatory Commission (NRC) estimated that the cooling tower system would affect plant efficiency by 3 percent (Gulvas, 2002). Consumers Energy estimates that the cooling system fans utilize 4 MW of electricity for operation, and the circulating and intake pumps utilize approximately 16 MW and 3 MW, respectively (Gulvas, 2002). Consumers Energy further estimate that the cooling tower system reduces the efficiency of the steam turbine by 6 to 8 percent compared to the original once-through system. Consumers Energy did not provide supporting documentation for the turbine efficiency penalties or pumping and fanning losses as submitted to the Agency.

Based on the Agency's energy penalty methodology, the turbine energy penalty for a nuclear unit (at peak summer conditions) would be approximately 1.4 percent (11.3 MW for Palisades). The Agency calculated this penalty using the historic cooling water temperature data for Palisades provided by Consumers Energy and ambient dry bulb and wet bulb air temperatures specific to Chicago, IL (Consumers, 2001).[1] This estimate of turbine efficiency penalty is substantially less than that estimated by Consumers Energy. The Agency notes that Consumers Energy did not estimate the original pumping requirements of the once-through system, and, therefore, the net energy penalty (that is, wet cooling tower energy use less the once-through system energy use) of the conversion estimated by Consumers Energy may not be the appropriate comparison. The Agency also notes that the electricity usage of 36-200 hp fans would be 5.4 MW with each fan at full operation, slightly higher than the estimate by Consumers Energy. EPA also estimated the pumping energy penalty for the recirculating system at Palisades and compared this to the pumping energy required for the former once-through operation of the plant. EPA determined, conservatively (that is, erring on the high side), that the circulating pumping requirements of the cooling tower system currently in place at Palisades would require approximately 7.5 MW (that is, 8.5 MW less than the estimate given by Palisades above). The original once-through system would have required approximately 5 MW to convey water 3,300 feet from the offshore intake through 11 ft diameter pipe, through the condenser, and to discharge at the lake shore. The Agency did not analyze the "dilution" pumping requirements estimated by Consumers Energy as 3 MW above. Therefore, the Agency estimates that the total energy penalty of the recirculating tower system at Palisades may have a peak energy penalty close to 2.7 percent and an annual penalty approaching 1.8 percent as compared to the original once-through system (Sunda, et al., 2002).

[1] The EPA calculations for energy penalties specific to Palisades and Lake Michigan utilized the data from the 2001 Consumers Energy permit document with the energy penalty methodology outlined in this chapter.

References

Air Force Combat Climatology Center in Ashville, NC. CD entitled Engineering Weather Data (2000 Interactive Edition). National Climatic Data Center.

Burns, J. M. and W. C. Micheletti. November 2000. Comparison of Wet and Dry Cooling Systems for Combined Cycle Power Plants. Version 2.1. Submitted as Appendix F to the comments of the Utility Water Act Group on EPA's Proposed Section 316(b) Rule for New Facilities.

Consumers Energy Company and Nuclear Management Corporation, May 2001, "Biological Assessment of the 1999 Cooling Water Flow Increase at the Palisades Nuclear Plant, near South Haven, Michigan."

Durand, A et al. May 1999. *Updated Rules for Pipe Sizing*. Chemical Engineering.

Entergy Nuclear Generation Company (Entergy). February 2001. *Condenser Performance Analysis - Additional Data; Pilgrim Nuclear Power Station.* Submitted by J.F. Alexander, to Nicholas Prodany, EPA Region 1.

Environmental Protection Agency. September 2001. *The Emissions & Generation Resource Integrated Database 2000 (E-GRID 2000).* Version 2.0. http://www.epa.gov/airmarkets/egrid/index.html

Fleming, Robert. 2001. Personal communications between Robert Fleming, The Marley Cooling Tower Co., and Faysal Bekdash, SAIC.

General Electric. No Date. Steam Turbine Technology. Field Engineering Development Center Mechanical & Nuclear.

Gulvas, J. A., 2002, "The Palisades Plant Once-through and Cooling Towers," Consumers Energy data submission to the U.S. EPA. February 28, 2002.

Heat Exchanger Systems, Inc (HES). 2001. Condenser Performance Analysis.

Hensley, J.C. 1985. *Cooling Tower Fundamentals.* 2nd Edition. The Marley Cooling Tower Company (Mission, Kansas).

Hess, Dale. June 2001. Condenser Cost Study. Graham Corporation.

Ishigai, S. 1999. *Steam Power Engineering - Thermal Hydraulic Design Principles.* Cambridge university Press. UK.

Kirk-Othmer. 1997. *Encyclopedia of Chemical Technology*. Fourth Edition. Volume 22. John Wiley and Sons, Inc. New York.

Litton, T.R. Stanley Consultants, Inc. No Date. "*Application of Parallel wet and Dry Condensing Systems to a 35 MW Steam Turbine*."

Mirsky, Gary. 2001. Personal communications between Gary Mirsky, Hamon Cooling Towers, and Faysal Bekdash, SAIC.

Nuclear Regulatory Commission. 1996. *Generic Environmental Impact Statement for License Renewal of Nuclear Plants*. NUREG-1437 Vol. 1. http://www.nrc.gov/docs/nuregs/staff/sr1437/V1/sr1437v1.html#_1_128

Perry, W. J., 1991, "Adverse Impact Study; Cooling Water Facilities - Jefferies Generating Station," for Santee Cooper, Moncks Corners, South Carolina.

Sunda, J.B., et al., 2002, "Analysis of Palisades Nuclear Plant Pumping Energy Requirements."

Tallon, B. Not Dated. GEA Power Systems Inc. Telephone Contact with John Sunda, SAIC. Regarding Air Cooled Condenser Fans.

Tatar, G. October 2001. Telephone Contact with John Sunda, SAIC. Regarding operation of the air cooled condenser fans. El Dorado Energy.

Taylor, S. May 2001. Telephone Contact with John Sunda, SAIC. Regarding cooling water pumping and condenser operation. Bechtel.

U.S. Department of Energy. May 2000. Statement of Jay E. Hakes Administrator, Energy Information Administration, Department of Energy before the Committee on Energy and Natural Resources, United States Senate.

U.S. Department of Energy. August 2001. "Energy Penalty Analysis on Possible Cooling Water Intake Structure Requirements on Existing Coal-Fired Power Plants," Working Draft, Office of Fossil Fuel Energy, National Energy Technology Laboratory, and Argonne National Laboratory.

U.S. Environmental Protection Agency. April 2001. Memorandum from Tim Connor to Tom Feeley (U.S. Department of Energy). "Comments on Proposed Energy Penalty Analysis."

Woodruff, E.B., Lammers, H.B., Lammers, T.F. 1998. *Steam Plant Operation.* Seventh Edition. McGraw-Hill. New York.